Abigail Daken, Manager
ENERGY STAR HVAC Program
US Environmental Protection Agency Office of Air and Radiation
1200 Pennsylvania Avenue NW
Washington, DC 20460

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Ms. Daken,

Northeast Energy Efficiency Partnerships (NEEP) appreciates the opportunity to provide comments to the ENERGY STAR program on the Residential Air Source Heat Pump and Central Air Conditioner V6.0 Specification Draft 2. NEEP launched a regional market transformation initiative for air-source heat pumps (ASHP) in 2013. One of the tools that came out of our initiative strategies was the development of the cold-climate air-source heat pump (ccASHP) Specification and product list. The specification development and evolution has been stakeholder driven and is now on its third version. Management of the Specification and product list has offered valuable perspective that is very relevant to this effort to update the ENERGY STAR Criteria for ASHPs, particularly with the inclusion of a new cold-climate category. As part of the management of the ccASHP Specification, NEEP maintains a sub-committee made up of representatives from efficiency programs, technical experts and the designer community to help inform needed changes. Several of the subcommittee members reviewed and provided input into this set of comments.

In general, NEEP is very supportive of this draft specification and feels it would positively contribute to identification of high performance ASHPs in the market.

We support the move to create a regionally-specific criteria. We came to a similar conclusion that there are different performance priorities between systems operating in cold climate applications from those operating in warm or moderate climate applications. In order to deliver real saving for consumers, the technical requirements should be customized for at least two regions. We believe it is useful for the criteria to align where possible with NEEP's ccASHP Specification in the interest of consistency and building off of an existing specification that has developed a significant level of market awareness and investment. The ccASHP Specification is on its 3rd version and has evolved through extensive stakeholder engagement input.

Comparison of ENERGY STAR Cold Climate category (Draft2) and the ccASHP Specification

	ccASHP Spec	ES V6.0 Draft 2 (Cold Climate)	ES V6.0 Draft 2 (Moderate and Hot)
HSPF	10/9 (ductless/ducted)	10*	9.2*
SEER	15	17*	17*
EER	No requirement	11.5*	12.5*
COP@5	1.75 (manufacturer reported)	1.75 (manufacturer reported until 2023, M1 required after)	
Capacity Maintenance	No requirement	70% (5F/47F) (manufacturer reported until 2023, M1 required after)	No requirement
Controls verification procedure*	No requirement	Required (TBD procedure)	No requirement
# of capacity stages	3+	2+	2+
Installation Capabilities	Option to report QI functionalities	Must meet 3 of 6 possible capabilities	Must meet 3 of 6 possible capabilities

*Early certification alternate metrics only for comparison

There are a few elements of the Draft 2 requirements that we want to provide more detailed comment including;

- Determination of Low Ambient performance
- Recommendation for moving from Percent of Heating Capacity (or "capacity maintenance") to an improved definition based on a "turn down" ratio requirement, including expanded use of a Controls Verification Procedure (CVP)
- The case for differentiating requirements for ducted products and ductless products

- Strong support for inclusion of required Quality Installation Capabilities

Low Ambient Performance:

Since its inception, the ccASHP has included a required COP of 1.75 at 5F. Understanding performance at this temperature provided assurances to designers and consumers in cold climates that a heat pump would be operating efficiently at temperatures that are typical of their design temperatures. Properly sizing ASHPs in cold climates requires knowing performance at these temperatures. From the launch of the ccASHP Specification in 2015, there has been an interest to base the low temperature performance requirement on industry standard test procedures. We support the adoption of the M1 test procedure to determine this performance as part of the ENERGY STAR Spec. We support the alignment of 5F COP levels with the ccASHP Spec level of 1.75. We also support allowing this performance to be manufacturer reported until 2023. We are very supportive of the inclusion of a CVP to confirm system ability to reach performance levels under native controls, a significant deficiency that permeates the current rating system.

EPA should also monitor progress of CSA's EXP-07 test procedure for variable capacity heat pumps. We support the potential incorporation of this test procedure after it is finalized, as a pathway to demonstrate performance according to ENERGY STAR requirements.

Percent of Heating Capacity

As part of NEEP's process to develop and update the ccASHP Specification, there has been a robust and ongoing conversation about the inclusion of a capacity maintenance requirement. We acknowledge that having accurate cold temperature capacity is really important. Some manufacturers raised concerns about gaming with respect to what some manufacturers call their "rated" capacity at 47, some intentionally picking lower capacity values which would help their low-ambient capacity look better in comparison. NEEP decided to address this by requiring transparent reporting over developing a strict cut off line. Armed with the information, a designer/installer can make a determination of whether a particular system meets the particular application needs. We don't believe any capacity maintenance ratio that includes the use of the rated capacity at 47F is particularly useful.

Alternatively, EPA should consider replacing the 70% capacity requirement with a requirement that is based on the ratio of the maximum capacity at 5°F to the *minimum* capacity at 47°F. To be useful, an additional CVP test that would address performance at low loads and mild temperatures (47°F) would be essential.

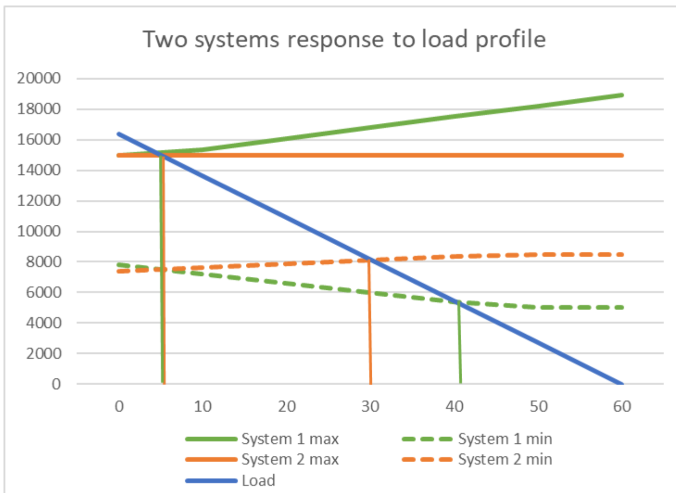
The colder a climate, the lower the design temperature, and the larger the capacity at design a heat pump needs to be. But the same heating load is there at mild temperatures, as it would in a milder climate; the colder a climate, the wider the range of modulation that is needed to ensure efficient operation. Building loads are roughly linear, and in a given building the load at 47°F is approximately 35% of the load at 5°F. If (for example) a variable speed system that has adequate capacity at 5°F can't modulate properly at very mild temperatures (say above 47), the kWh penalty is relatively small because there's not that much total heating energy at those temperatures. But the colder the temperature at which the compressor stops modulating and begins to cycle, the further and faster the annual efficiency drops. If the ratio of capacity at 5°F at high speed to the capacity at

47°F at low speed is much smaller than 2.8:1 (and the system is sized to exactly meet the load at 5°F), then the system will be guaranteed to cycle at temperatures below 47°F. If the system is actually sized for a lower design temperature, or is oversized at 5°F, that cycling temperature will be even lower.

An illustrative example follows. Consider two systems; system 1 has a higher turn-down ratio (lower 47F minimum capacity) even though it has a much lower "capacity maintenance" value:

Capacities (Btu/h)	47F rated	47F min	5F max	turn-down ratio	"capacity maintenance"
System 1	18000	5000	15000	3	83%
System 2	15000	8500	15000	1.8	100%

The maximum (solid) and minimum (dotted) capacities of both machines are plotted against the load line (blue) of a home for which both are sized to meet the design load at 5°F:



The most important element in the maximum capacity plot is where it intersects with the load line; any outdoor temperatures below that intersection will need auxiliary heat or will suffer a temperature drop in the home. For virtually all temperatures above that point, the system will presumably be modulating, until the temperature reaches the point where the minimum capacity curve intersects the load line. Above that temperature, the equipment will not be able to modulate and will cycle, leading to decreased efficiency.

In the example, System 1 has a high turn-down (3) and a lower "capacity maintenance" value; its "cycling limit" is about 42°F. System 3 has a much lower turn-down (1.8), even though its "capacity maintenance" is higher; but the cycling limit temperature is much colder, about 30°F. This system will operate at lower efficiencies for a significant part of the winter (about half of heating hours, assuming the average heating temperature is about 30°F). This suggests that a well-reported (reliable) turn-down ratio is far more important to ensure heating performance over a wide range of temperatures than a simple "capacity maintenance" value. This effect is amplified in cases of oversizing, and further amplified if low-speed performance at 47°F is not achievable in the field.

CVP test: Although a CVP test is useful to confirm performance at 5°F, it's also relatively likely that manufacturers built-in control algorithms will provide the rated performance (or close to it) at such cold temperatures, because they have a very big incentive to ensure that their equipment will provide the stated heating capacities at or near typical design conditions; the downstream consequences of not meeting heating loads is severe among distributors, engineers, dealer/installers, and end-use consumers alike. The most likely control algorithm that might limit heating capacity at 5°F would likely be in place only to ensure that equipment longevity doesn't drop due to higher compressor speeds, a trade-off that manufacturers are likely extremely aware of during design and rating testing.

At 47°F, however, there is little incentive to ensure that *low-speed* performance that is reported is actually achievable under normal controls. There is considerable evidence from on-site monitoring and other sources that many products repeatedly cycle at much higher power levels than their stated minimum- continuous operating levels (whether stated in engineering specifications or reported on the ccASHP cold-climate listings, or both). It is likely that some combination of control algorithms and thermostat sensing strategy contributes to the discrepancy; at least one manufacturer successfully introduced a firmware change several years ago that dramatically enhanced the mild-weather heating efficiency of one of their product lines.

The origins of the CVP test were in validating part-load performance of VRF systems, and the same approach would apply here. Appendix M1 already includes a mandatory 47°F low-speed test ($H1^1$), using fixed compressor frequencies and fan speeds; a CVP test that validated such performance would dramatically improve assurance that the reported low-speed data is realistic. And tying the high-speed capacity at 5°F to that *validated* low-speed performance at 47°F would be vastly more useful for ensuring performance over the wide range that is needed in cold climates, as well as providing improved data for designers and engineers to use in sizing systems.

Creating new categories (Ducted and Ductless) with different HSPF levels

We believe there is justification for differentiating required HSPF levels for products that utilize compact ducts and centrally ducted products, due to differences in testing conditions used to test these systems (per AHRI 210/240).

Ducted systems require more energy than ductless systems to deliver the same amount of space conditioning through the ducts, because of the additional friction of the duct system. Although the ratings of similar ductless products (units that otherwise have the same ratings, controls, and utilize the same outdoor models) would seem to indicate that they are 20-25% more efficient than ducted models, researchers from NREL/PNNL estimate the real differences in performance from the added fan energy to be about 4%[1]. This value was confirmed independently by Proctor Engineering Group[2]. Subtracting 4% from the 20-25% ratings difference still leaves a 15-20% gap in the efficiency ratings between ductless and ducted systems that are otherwise nearly identical.

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[1] C.E. METZGER, J. ZHANG, PH.D., J. MAGUIRE; J. WINKLER, Are Ducted Mini-Splits Worth It? , ASHRAE Journal, February 2018

[2] Personal communication from Abram Conant, Proctor Engineering Group, 10/18/2017: "… the EER difference due to higher fan energy for the ducted head would be on the order of ~4% for the units we tested last year."

Non-ducted systems are tested under conditions that appear to report performance differently compared to those that have ducts, because the ductless systems can be operated at higher fan speeds during the rating tests. While there is a maximum air flow specified in the test procedure for ducted systems (equivalent to 444 CFM/ton), there is no maximum specified for ductless systems during the test. The increased fan power during testing is small compared to the increased capacity (and therefore efficiency) produced by higher flow rates, even though ductless fan coils would never be run at “test mode” fan speeds during normal operation. This test condition would result in a larger discrepancy in rated performance results that are not borne out in the field.

In addition, we believe there is significant, cost-effective savings to consumers by encouraging compact-ducted systems (typically mid-static air handlers with ducts leading to 2-4 small rooms) over multi-zone individual ductless in each room. Though the installation cost would typically be similar, installing multiple zones that are oversized into multiple rooms with small loads exacerbates excessive compressor cycling and lower efficiencies under lower-load conditions. Each zone installed in a small room that is oversized leads to a larger and larger outdoor unit, yet the larger outdoor units have smaller turn-down ratios. Under part-load conditions the minimum capacity of the unit may exceed the total house load by several times, leading to excessive cycling. Two likely solutions to improve performance would be by installing more single-zone systems (that each have larger individual turn-down ratios), and by combining several small rooms into a shared zones by using a compact-ducted system. The option to offer fewer wall-mounted ductless terminals and/or fewer outdoor units provides more options for designers and installers to offer alternative solutions that can meet customers’ needs. Having ducted heat pump systems that are listed (and that may in the long run be as efficient, if not more so, than their ductless counterparts) is an important consideration.

For these reasons, NEEP adopted a lower HSPF requirement for ducted products in version 3.0 of the ccASHP Specification. We suggest ENERGY STAR consider separating these groups of products to reflect a slightly more stringent requirement for ductless systems compared to ducted and mixed systems.

Quality Installation functionalities

High-quality installations of air-source heat pump (ASHP) systems generate referrals, increase sales, reduce callbacks and improve customer comfort and satisfaction. Installation practices also have a major impact on efficiency and performance of an ASHP system. We applaud ENERGY STAR for including functionalities that help ensure that these high performance systems operate to their potential.

Effective Date

We support the timing of the proposed effective date of January 2023 and support the ability for manufacturers to certify products to V6.0 as soon as the specification is finalized.

Conclusion

Thank you for offering the opportunity for NEEP to provide comment to the ASAHP/CAC V6.0 draft 2 specification. ENERGY STAR must continue to serve in a leading role in recognition of high performing energy efficient products, and NEEP looks forward to continuing to support ENERGY STAR’s efforts into the future. Please don’t hesitate to contact us with any follow up questions or clarifications.

Sincerely,

David Lis
Director of Technology and Market Solutions
Northeast Energy Efficiency Partnerships (NEEP)
djlis@neep.org
781-860-9177 x127